

An experimental study on the performance of the moving regenerator for a γ -type twin power piston Stirling engine



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ABSTRACT

In this paper, a helium charge γ -type twin power piston Stirling engine has been studied experimentally to understand the effects of several regenerator parameters on the overall performance of the engine. The regenerator incorporated in this engine is a moving regenerator which is housed inside the displacer of the engine, and the parameters investigated include regenerator matrix material, matrices arrangement, matrix wire diameter, and fill factor. Stacked-woven metal screens have been used as regenerator matrix materials. The results include engine shaft torque, power, and efficiency versus engine speed at several engine's hot-end temperatures. It is found that all parameters pose significant impact on engine performance. Copper is a superior regenerator material than stainless steel for the current engine; regenerator matrix screens have to be installed in a manner that the working-gas-flow direction is normal to the surface of matrix screens; very small wire diameter results in large pressure drop and reduce regenerator effectiveness; and there exists an optimal fill factor. The study offers some important information for the design of moving regenerator in a γ -type Stirling engine.

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1. Introduction

Among the possible solutions to address today's energy and environmental problems, Stirling engines have attracted much attention in recent years because their advantages of high thermal efficiency, low maintenance requirement, flexibility on energy sources, and safe to operate. In particular, their ability to use a wide range of energy sources such as solar energy, industrial waste heat, and biomass fuel has made them one of the most promising answers to the worsening problems of fossil fuel depletion and global warming. Now Stirling engines have been the focus of many academic studies, both experimentally and numerically [1–3], and a new industry centered on Stirling engines and their applications has emerged [4]. There exist many varieties of Stirling engines; and these are generally classified into three different catalogues according to their configurations, namely α , β , and γ types. Regardless the different engine types, a regenerator is a necessity for a Stirling engine to be efficient. It improves engine's thermal efficiency by functioning as a thermal heat storage unit. When hot gas flows through it in the expansion process of an engine cycle, it absorbs part of the heat from hot gas and then release the heat back to cold gas as it flows through the regenerator in the compression process. Consequently, less heat input is needed at the hot end of the engine, hence improving engine's thermal efficiency. Tavakolpour et al. [5]

studied a LTD (low temperature difference) Stirling engine numerically and found that the engine's thermal efficiency resulted from regenerator effectiveness of 1.0 is 6 times larger than that resulted from zero regenerator effectiveness. A regenerator is basically a void filled with porous medium made of metal felt or matrices. The combined surfaces of the felt or matrix wires create a large heat transfer area that promotes heat transfer rate between working gas and the porous medium, and the metal material itself provides huge heat capacity to absorb and release large amount of heat energy. However, there exist disadvantages associated with a regenerator. The void in regenerator increases the dead volume of the engine, and the porous medium structure inside the regenerator increases pressure drop as working gas flows forwards and backwards through the regenerator in an engine cycle. In addition, there are energy losses in the regenerator due to temperature oscillation and heat conduction caused by the alternating gas flow and steep temperature gradient throughout the regenerator. A well performing regenerator needs to maximize heat transfer rate and heat capacity while minimize dead volume, pressure drop, and energy losses. Therefore, regenerator optimization is at the heart of Stirling engine design [6].

There are different ways of incorporating a regenerator into a Stirling engine. The regenerator space can be constructed by adding an outer shell upon the outer wall of displacer cylinder to create an annular space [7] or simply by using the existing space inside the displacer itself [8]. In the former, the regenerator is stationary, while in the latter, the regenerator moves up and down

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Nomenclature

c_p	constant pressure specific heat ($\text{kJ kg}^{-1} \text{K}^{-1}$)	R_λ	thermal conductivity ratio between solid and gas materials
d	regenerator matrix screen wire diameter (m)	r_1	crank radius of the power piston (m)
f	fill factor	r_2	crank radius of the displacer (m)
k	thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)	T	temperature (K)
l_1	length of the power piston linkage bar (m)	$\dot{W}$	engine power (W)
l_2	length of the power piston connection rod (m)		
l_3	length of the displacer linkage bar (m)	Greeks	
l_4	length of the displacer connection rod (m)	η	engine efficiency
l_d	height of displacer (m)	θ_p	phase angle between displacer and power piston
m	mass (kg)	τ	engine shaft torque (N-m)
P_0	initial charge pressure (Pa)	ω	engine speed (rad s^{-1})
$\dot{Q}$	hot-end heat transfer rate (W)		
R_1	outer radius of the power piston (m)	Subscripts	
R_2	inner radius of the displacer cylinder (m)	h	hot end
R_d	outer radius of the displacer (m)	l	cold end

with the displacer, hence the latter is termed ‘moving regenerator’ in this study. There are advantages and disadvantages for both designs. The stationary regenerator is more complicated to construct, however, there is little limitation on the weight of the regenerator matrix since it is not moving. The moving regenerator design is based on the principle of multi-functional capability of parts because the displacer can also function as the regenerator. In this design, no additional construction is needed to create the extra space for regenerator; therefore it is much simpler to build. However, the design puts more weight on the displacer, and this, in turn, gives rise to larger momentum change during the reciprocation motion of the displacer and creates more friction and vibration on the crank shaft. Consequently, this problem, to a certain degree, limits the amount of matrix material that can be put inside the regenerator, hence limiting the regenerator’s heat capacity. Furthermore, how to secure the regenerator matrix inside the displacer to withstand rapid reciprocation motion of the displacer becomes an issue. These two problems get worse when the engine runs at high speed. It seems that there are more problems related to the moving regenerator design; hence the stationary regenerator is favored by engineers and has been widely adopted in most Stirling engines. There have been many reports on the performance of the stationary regenerators. Costa et al. [6] presented a numerical study on the pressure drop phenomena in a wound woven wire matrix of a Stirling regenerator. Several correlations have been proposed to characterize pressure drop friction factor for wound woven wire regenerator matrices, and these can be very helpful to obtain more accurate numerical prediction on engine performance. However, heat transfer characteristics in regenerator have not been investigated in the study. Andersen et al. [7] conducted a detailed numerical study on the regenerator of a SM5 α -type Stirling engine [9]. They investigated the effects of two parameters, regenerator matrix wire diameter and fill factor, on electric power, efficiency, regenerator loss, and heat intake. Here, fill factor indicates the fraction of volume inside the regenerator that has been occupied by matrix material. This parameter is closely associated with pressure drop of working gas as it flows through the regenerator. Particular interest has been focused on the impact of matrix temperature oscillation on regenerator performance. It was concluded that regarding the two parameters, the current regenerator design of the SM5 engine is close to the optimal point. They also found that power output decreased sharply for low fill factors and large wire diameters, which resulted in smaller total heat transfer area in the regenerator matrix. A decrease in power also occurred for very

small wire diameters and large fill factors, which resulted in larger pressure drop across the regenerator matrix. The study demonstrated the importance of optimizing the regenerator matrix on the overall performance of a Stirling engine. Bin-Nun and Maniatis [10] designed and tested a laminate screen matrix regenerator with 47 elements. A laminate sintering process was used to bind the stainless steel screens together to form strong regenerator matrix. The regenerator was tested in a Stirling refrigerator where regenerator effectiveness plays a very crucial role to refrigeration efficiency. It was shown that a mere reduction of regenerator effectiveness from 100% to 99% resulted in a loss of 20% in refrigeration power in their Stirling refrigerator under certain conditions. In terms of the moving regenerator, there have been very few investigations on its performance in open literature. Kongtragool and Wongwises [8] conducted an experimental study on low-temperature difference γ -type Stirling engines. Two engines, one with 2 power pistons and the other with four power pistons, were investigated, and both engines adopted moving regenerators. The results included variations of engine torque, shaft power, and engine’s thermal efficiency versus heat input. However, there is little information regarding the impact of the effectiveness of the moving regenerators on engine performance.

Although the parameters of matrix wire diameter and fill factor examined in [7] are closely associated with heat transfer area, pressure drop, and heat capacity of the regenerator, they do not totally address the heat transfer rate between the regenerator matrices and working gas. Since working gas flows through the regenerator, the dominate heat transfer mechanism between regenerator matrices and working gas is convective heat transfer. In this perspective, the heat transfer rate of regenerator matrices is strongly dictated by the arrangement of the matrices themselves, that is, how matrix wires are placed relatively to the direction of gas flow. Considering a single matrix wire, if the axis of the matrix wire is perpendicular to the gas flow direction, a scenario of impingement flow is created where the heat transfer rate between the wire and the gas reaches maximum. On the other hand, if the axis of the wire is parallel to the gas flow direction, a scenario similar to poiseuille flow is returned, and the heat transfer rate reaches minimum. Therefore, randomly placed matrices (such as felt) and well-arranged matrices (such as wire screen) would result in different heat transfer rates in a regenerator even though the wire diameter and fill factor are the same. Regenerator heat transfer rate is a very crucial factor to regenerator effectiveness especially at high engine speed when the time for heat transfer is very short

in each cycle. Therefore, a regenerator with low heat transfer rate will perform poorly at high engine speed even though it has large heat capacity. Based on the above argument, a comprehensive parametric study on regenerator matrix should take the arrangement of regenerator matrices into account.

As argued in [7] that the effects of regenerator matrix parameters on engine performance observed in their study will likely be different with those on engines in different designs, likewise the results obtained for a stationary regenerator do not necessarily apply to a moving regenerator. Yet, there has not been an in-depth study on the performance of moving regenerator in open literature. There are gaps to be filled in this aspect. Therefore, the objective of this study is to experimentally examine the impacts of several parameters on the performance of the moving regenerator. These parameters include matrix mesh material, matrix arrangement in relation to gas flow direction, matrix screen wire diameter, and fill factor. The engine selected in the current investigation is a helium charged γ -type twin power piston Stirling engine in Chen et al. [11]. The γ -type engine is chosen because of its many specific advantages described in Kongtragool and Wongwises [12]. It is a low to medium temperature differential engine; therefore it can run on temperature difference from just tens of degree to a couple of hundreds of degree. This is a very important advantage because the engine can be powered by a wide range of energy sources, including low-grade sources such as solar, biofuel, geothermal, or even industrial waste heat. It is also much easier and cheaper to be constructed and maintained than other types of Stirling engines. Therefore, the γ -type engine is very suitable for domestic use, which can potentially become a very large market, because of the readily availability and low cost of low-grade fuel and its simplicity to construct and maintain. In the numerical study of [11], the performance of the regenerator was only represented by constant regenerator effectiveness. Although they reported that regenerator effectiveness is the most important factor to promote the engine's thermal efficiency, what regenerator parameters and to what extent their contributions to regenerator effectiveness and engine performance have not been examined. These are very important issues for a real engine to be built. This paper is a followed-up study of [11] to fill the gaps and to shed some light on these crucial problems.

2. Experimental apparatus

The details of design and construction of the current γ -type Stirling engine have been documented in [11] and will not be repeated here. The schematic diagram of the test facility, including a γ -type Stirling engine, magnetic couplings, torque transducer, transmission gears and chain, loading, and photo tachometer, is illustrated in Fig. 1. A photograph of the test facility is given in Fig. 2. The Stirling engine is heated by a 2 kW electric heater wrapped around the lower portion of the displacer cylinder (engine's hot end) and is cooled by a water jacket installed at the upper portion of the displacer cylinder (engine's cold end). The heater is capable of heating the engine's hot end up to 773 K, and its heating power is controllable by an electric power regulator. This allows the electric power supplied to the heater to be fixed at a certain desired level. A thermal couple, with the accuracy of ± 0.1 K, is installed at the hot end to measure the hot-end temperature. The water at the cold-end water jacket is circulated by a water pump, which pumps water from a large external water tank where the water temperature is maintained at around 298 K. Since the entire engine, including the crank case, is sealed inside a pressure vessel, the engine's power is transmitted to an external power shaft via a pair of magnetic couplings which are capable of handling torque up to 4 N-m. The power shaft is mounted on a xyz sliding platform to facilitate

accurate alignment of the magnetic couplings. A torque transducer, model TQ-25R by KOTSAO and with the accuracy of ± 0.01 N-m, is installed at one end of the power shaft to serve as the connection of the power shaft to a loading gear and to measure the torque on the power shaft. The loading gear drives a chain which in turn drives a smaller gear installed at a loading shaft. At one end of the loading shaft, an 18-in 4-blade fan is attached to produce the loading, and at the other end, a flywheel is attached to stabilize the rotation speed of the loading shaft. A photo tachometer, model UT-371 by TECPEL and with the accuracy of ± 0.01 RPM, is positioned in front of the fan to measure its rotation speed. The engine's rotation speed can be deduced by dividing the fan's speed by the gear ratio between the large and the small loading gears. With the torque of power shaft and engine speed available, the engine's output power can be calculated by:

$$\dot{W} = \tau \times \omega \quad (1)$$

where τ is torque, and ω is engine speed in radian per second. To evaluate engine's power output under different loadings, there has to be some degree of control on the magnitude of loading. This can be done by changing the gear ratio between the large and small loading gears. That is, the larger the gear ratio, the larger the loading on the engine. In this study, the number of teeth of the small loading gear is fixed at 13, whereas, there are 7 different large loading gears whose teeth numbers are 30, 36, 40, 45, 50, 55, and 60, respectively. The combinations of these large gears and the small gear give rise to gear ratios of 2.37, 2.77, 3.07, 3.46, 3.84, 4.23, and 4.62, respectively. To evaluate engine's efficiency, the information on the amount of heat input at the hot-end in each cycle is needed. We estimate heat input by using the numerical code developed in [11]. Here, some parameters in the code, such as the convective heat transfer coefficients in the expansion and compression chambers and the regenerator effectiveness, are tuned to match the measured power output and engine speed, and the resulted heat input from the numerical simulation is used as the heat input to evaluate engine's efficiency. The heat transfer rate at hot end can be obtained by multiplying heat input to engine revolution per second. With the hot-end heat transfer rate available, the engine efficiency can be evaluated by:

$$\eta = \frac{\dot{W}}{\dot{Q}} \quad (2)$$

3. Experimental procedure

To investigate the effects of regenerator parameters, an experimental cycle has been run whenever the regenerator is changed. In each experimental cycle, a large loading gear, starting with the one with the least teeth, is selected first. The engine is initially heated with full electric power and then hand started when the hot-end reaches a certain temperature, normally around 423 K. When the hot-end temperature further rises to a certain desired level, say 473 K, the electric power supplied to the heater is dialed down to maintain the hot-end temperature at that desired level. After careful adjustment on the heating power, the engine can run at a steady state, and the measurements on fan speed and power shaft torque can be taken. Then the heating power is dialed up again until the hot-end temperature reaches the next desired level, and another attempt on stabilizing hot-end temperature and taking measurement is performed. The process of heating, stabilizing, and taking measurement is repeated over several different hot-end temperatures to complete a run on that particular loading gear. At the end of a run, the heater is powered off, and the engine is left to be cooled back to room temperature. Then the large loading gear is changed to another one with a different teeth

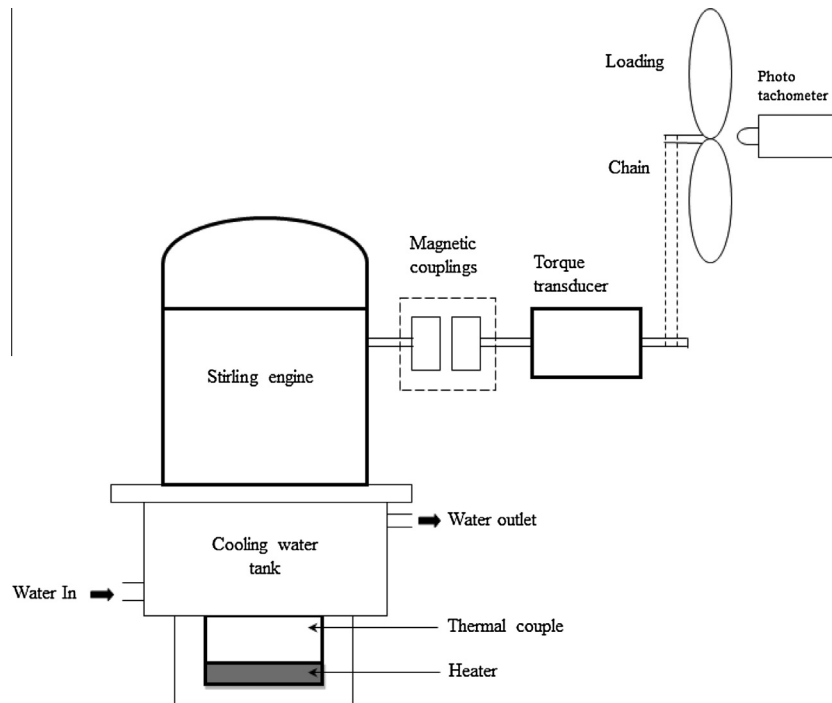


Fig. 1. Schematic of experimental apparatus.

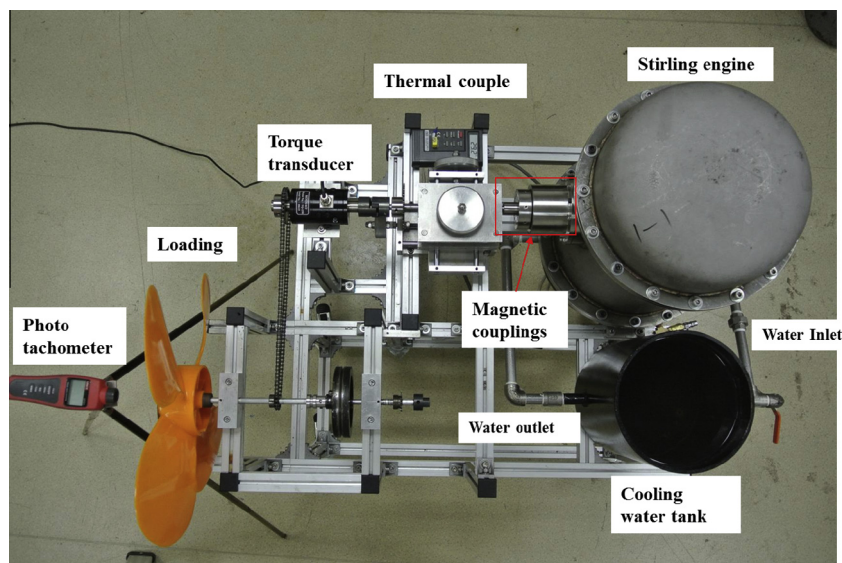


Fig. 2. Top view of the experimental apparatus.

number, and the whole procedure repeats again. After all large loading gears are tested; the engine is disassembled to replace a new moving regenerator, and another cycle of experiment begins.

4. Results and discussion

The objective of this study is to investigate the effects of several regenerator parameters on engine performance. Therefore, the geometric parameters of the engine, including strokes of the power piston and displacer, compression ratio and displacer length, are all fixed. A schematic diagram of the geometric parameters in this Stirling engine is given in Fig. 3, and their values are listed in Table 1. The working gas is helium, and the initial charge pressure is fixed at $P_0 = 0.3$ MPa (absolute pressure). To examine the performance

of the regenerator and the engine at different hot-end temperatures, the power to the electric heater is controlled to maintain the hot-end temperatures at 473 K, 523 K, 573 K, 623 K, and 673 K, respectively. The regenerator matrices used are metal wire screens. Metal felt made of very thin wires has been tested before, but without proper bracing, the felt was quickly compressed and fell to the bottom of the displacer under rapid reciprocation motion of the displacer. Metal wire screens, on the other hand, can be braced inside the displacer with much ease. Fig. 4 shows the explosion diagram and full assembly of the current moving regenerator. As can be seen in Fig. 4(a), the metal screens are cut into circular disks. Their centers are secured to the displacer shaft by some spacers (Fig. 4(b)), and their edges are glued to the inner wall of the displacer's lateral panel using temperature-resistance silicon

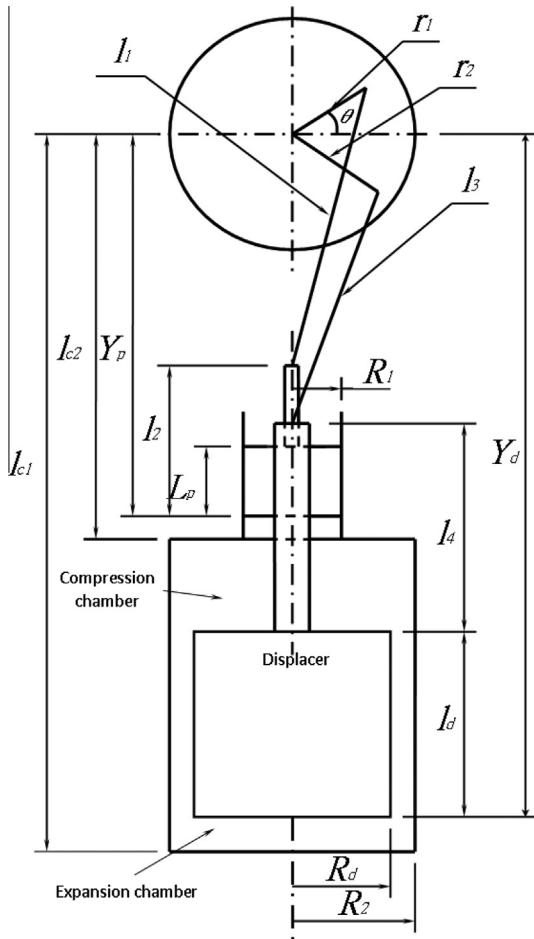


Fig. 3. Schematic of engine's geometric parameters.

Table 1
The values of the geometrical parameters of the baseline Stirling engine.

R_1 (m)	0.0256
R_2 (m)	0.0800
R_d (m)	0.0780
r_1 (m)	0.0400
r_2 (m)	0.0250
l_1 (m)	0.1300
l_2 (m)	0.0450
l_3 (m)	0.0950
l_4 (m)	0.1550
l_d (m)	0.1480
l_{c1} (m)	0.4250
l_{c2} (m)	0.2210
t (m)	0.0030
θ_p ($^\circ$)	90

glue. A moving regenerator constructed this way has been tested under engine speed of 1000 rpm without any dislocation of the metal screens. Also seen in Fig. 4(a), several holes have been drilled on the top and bottom plates of the displacer to allow part of the working gas to flow in and out of the regenerator space. One major advantage of this regenerator matrix assembly is that the general flow direction of the working gas is perpendicular to the matrix screen, creating impingement heat transfer between the matrix material and the working gas. Impingement heat transfer is known to achieve the highest heat transfer rate among mechanisms of convective heat transfer. Therefore, this design can give rise to a high heat transfer rate between matrix material and working gas.

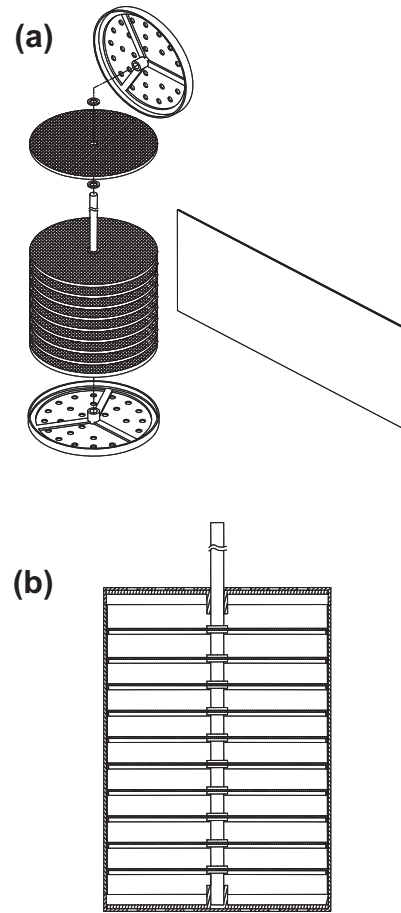


Fig. 4. Regenerator assembly, (a) explosion diagram and (b) full assembly.

Another advantage is that the axial thermal conductivity in matrix material is very low because all layers of matrix screens are separated by spacers and free from direct contact of each other.

The regenerator matrices used here are stacked-woven wire screens made of stainless steel or copper. They are woven in a pattern to form many tiny square holes that allow gas to pass through. The fineness scale of the wire screen is defined by the number of wires in an inch. For example, a 180-scale screen has 180 wires within the length of an inch in one direction. Fig. 5 shows microscopic photos of a 120-scale copper screen, a 180-scale copper screen, and a 180-scale stainless steel screen. Note that the unit of length scale in these pictures is micro meter. In this study, wire screen with 3 different fineness scales have been tested, namely 80, 120, and 180. The corresponding wire diameters are 0.00011 m, 0.00009 m and 0.00005 m, respectively. The regenerator parameters examined in this study include matrix material, matrix screen arrangement, screen wire diameter, and fill factor.

4.1. Effects of matrix material

Two different matrix materials have been tested, namely stainless steel and copper. The material properties important to moving regenerator effectiveness, including density, thermal conductivity, and specific heat, of the two materials are given in Table 2. Among the three properties, thermal conductivity of matrix material poses profound impact on convective heat transfer. Lopez Penha et al. [13] conducted a numerical study on fully-developed conjugate heat transfer in porous medium. They reported strong dependency of Nusselt number on R_x , the thermal conductivity ratio between

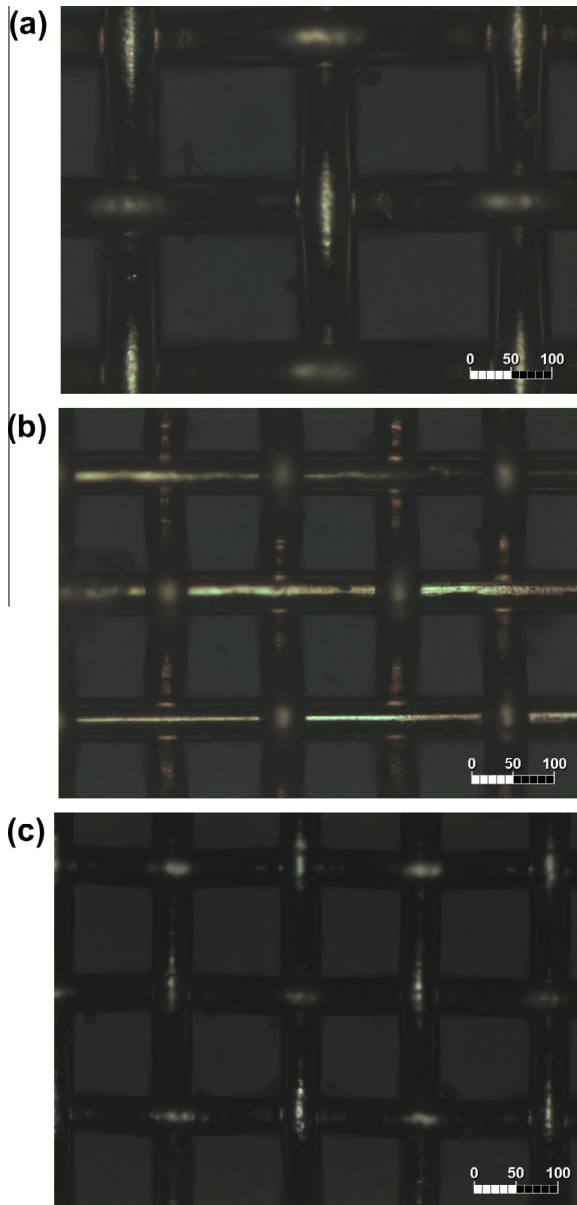


Fig. 5. Microscopic images of the stacked woven metal screens, (a) a 120-scale copper screen, (b) a 180-scale copper screen, and (c) a 180-scale stainless steel screen.

Table 2
Properties of stainless steel and copper.

	Stainless steel	Copper
Density (kg m^{-3})	7850	8960
Thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)	16	400
Specific heat ($\text{J kg}^{-1} \text{K}^{-1}$)	500	385

solid and gas media, up to $R_z = 200$ at a wide range of wire Reynolds numbers, from $\text{Re} = 1$ to 100. Within this range, the larger the value of R_z , the larger the Nusselt number. Comparing the properties between the two metal materials, copper is a bit denser, about 14%, than stainless steel; the thermal conductivity of copper is much larger, about 25 times, than stainless steel; but the specific heat of stainless steel is larger, about 30%, than copper. Although the value of helium thermal conductivity depends on pressure and temperature, within the conditions inside the current Stirling

engine, the values of R_z for stainless steel and copper are within the range from 10 to 200, respectively. Therefore thermal conductivity poses some impact on the convective heat transfer between matrix material and working gas in regenerator. Judging from the material properties, using stainless steel as matrix material yields slightly higher heat capacity, but using copper can produce much higher heat transfer rate. Since both heat capacity and heat transfer rate are very important to regenerator performance, the weighting of these two parameters on engine performance can only be realized through experiment.

The regenerator parameters, including matrix material, screen fineness scale, mass, number of discs, and fill factor, of all test cases in this study are listed in Table 3. In these, cases 1, 2, and 3 are designed to investigate the difference in performance resulted by using stainless steel or copper as matrix material. In cases 1 and 2, the fineness scale of metal screen and the number of screen discs are the same, but matrix material is different (case 1 is stainless steel, and case 2 is copper). This results in identical fill factor but different matrix mass. However, the heat capacity of matrix material in case 1 is 41.0 J K^{-1} , while that in case 2 is 38.5 J K^{-1} . That is, both cases have similar heat capacity. In case 3, the matrix material is copper, but the number of screen discs is decreased to 11 to reduce the matrix mass to almost the same as that of case 1. In brief summary, between cases 1 and 2, fill factor and heat capacity are controlled; while between cases 1 and 3, matrix mass is controlled. Fig. 6 shows the shaft torque, engine power, and engine efficiency of case 1 at 5 different hot-end temperatures. In Fig. 6(a), it can be seen that an increase in engine speed results in a decrease in the shaft torque. In Fig. 6(b) and (c), the engine power and efficiency have very similar tendency; they, in general, first increases with increasing engine speed, reaches a maximum value, and then decreases with increasing engine speed. The maximum engine power and efficiency are respectively 59.14 W and 5.35% at $T_h = 673 \text{ K}$ and $\omega = 515 \text{ rpm}$. In terms of temperature effect, Fig. 6 indicates that the magnitudes of the three quantities are almost proportional to hot-end temperature. The above are typical phenomena associated with the performance of a Stirling engine. Fig. 7 shows the shaft torque, engine power, and efficiency of cases 1, 2, and 3 at two different hot-end temperatures, 473 K and 673 K, which are respectively the lowest and highest temperatures among the five temperatures investigated here. It can be seen that generally the engine performance in case 3 is the best, whereas, case 1 is the worst. The shaft torque, engine power, and efficiency in case 2 are about 10% higher than those in case 1, and those in case 3 are about 20–30% higher than case 1. Since the regenerators in case 1 and 2 have the same fill factor and similar heat capacity, the superior engine performance of case 2 over case 1 should be rooted in the much higher thermal conductivity of copper than steel, which produces higher heat transfer rate. Comparing cases 1 and 3, both have almost the same matrix mass, but the matrix heat capacity of case 1 is 41.0 J K^{-1} , and that in case 3 is 30.8 J K^{-1} . Despite the lower matrix heat capacity in case 3, the engine still performs much better in case 3 (uses copper) than in case 1 (uses stainless steel). The test results in this group of cases clearly indicate that copper is a superior matrix material than stainless steel, and the physical mechanism is mainly due to the higher heat transfer rate produced by copper that is rooted in copper's much higher thermal conductivity than that of stainless steel. Although copper performs better in the current engine, this does not necessarily implies that it always performs better in other Stirling engines with different regenerator designs. Since copper is a better regenerator matrix material than stainless steel for the current Stirling engine, it is used for the rest of tests in this study.

Comparing cases 2 and 3, both use copper as matrix material, but case 3 has four less copper discs than case 2, hence its regenerator matrix has smaller heat capacity and fill factor than case

Table 3
Regenerator parameters in all test cases.

Case no.	Material	Screen fineness	Matrix mass (kg)	Disc number	Fill factor	R_d (m)
1	Stainless steel	180	0.082	15	0.00282	0.078
2	Copper	180	0.100	15	0.00282	0.078
3	Copper	180	0.080	11	0.00206	0.078
4	Copper	80	0.100	15	0.00282	0.078
5	Copper	80	0.100	–	0.00282	0.078
6	Copper	120	0.100	15	0.00282	0.078
7	Copper	80	0.072	11	0.00183	0.078
8	Copper	80	0.121	18	0.00356	0.078
9	Copper	80	0.100	15	0.00282	0.079

2. The smaller heat capacity is a disadvantage, but the smaller fill factor is an advantage because the pressure drop in the regenerator would be smaller. Since the engine performance in case 3 is better than case 2, it is reasonable to argue that in this circumstance, the advantage of smaller pressure drop outweighs the disadvantage of smaller heat capacity. The effect of fill factor will be further investigated in later sections.

4.2. Effects of matrix screen arrangement

Cases 4 and 5 are designed to investigate the effects of matrix screen arrangement. Both cases use the same copper screen, with fineness scale of 80, and the same amount of screen material. Hence, both have the same fill factor. Case 4 adopts typical installation of the regenerator matrix screens described earlier, creating impingement heat transfer between matrix screens and working gas. In case 5, on the other hand, a large sheet of matrix screen was first alternatively folded to create small creases and then rolled into forming a matrix cylinder. It was then placed inside the displacer with the displacer shaft passing through the center of the matrix cylinder. In this assembly, the axial directions of both matrix cylinder and displacer cylinder are parallel, so the general flow direction of the working gas is also parallel to the surface of matrix screen, resulting in the worst convective heat transfer scenario between matrix screens and working gas. Since all regenerator parameters, except matrix screen arrangement, in both cases are the same, any difference in engine performance can be attributed to the difference in how matrix screens are arranged relatively to the working gas flow. Fig. 8 illustrates the shaft torque, engine power, and efficiency of cases 4 and 5 at two different hot-end temperatures, $T_h = 473$ K and 673 K. The results of the engine without installing any matrix screen have also been given in Fig. 8 to serve as references to show the magnitudes of improvement by adopting regenerator matrices in cases 4 and 5. However, only the data at $T_h = 673$ K are available for the case without regenerator matrix because the engine wouldn't start at the lower temperature. It is noticeable that the magnitudes of all quantities in case 4 are significantly higher than those in case 5. The measurements in the former are about 230–250% larger than those in the latter. In comparison with results from the engine without regenerator matrix, the magnitudes of all quantities of cases 4 and 5 are respectively about 450% and 200% larger. Even at 673 K hot-end temperature, the efficiency of the engine without a regenerator matrix is only around 1%. The results clearly demonstrate the importance of incorporating regenerator matrices and the magnitude of difference in the effects on engine performance due to the arrangement of regenerator matrix screens. It also proves that convective heat transfer rate between matrix materials and working gas plays a significant role on the overall performance of the engine and is the major physical mechanism contributing to higher output power and efficiency in this group of cases. Therefore, the effect of matrix material arrangement cannot be overlooked in any study on regenerator performance.

4.3. Effects of matrix screen wire diameter (fineness scale)

Cases 2, 6, and 4 are designed to investigate the effects of matrix wire diameter (or fineness scale). The wire diameters are 0.00005 m, 0.00009 m, and 0.00011 m, respectively, and the screen fineness scales of the three cases are 180, 120, and 80, respectively. In all cases, the mass of matrix material is the same, giving rise to the same fill factor. A smaller wire diameter results in an advantage of larger total heat transfer area between matrix material and working gas, but it also results in a disadvantage of larger pressure drop because of the tighter weave of the wires. Fig. 9 shows the shaft torque, engine power, and efficiency of the three cases at two different hot-end temperatures, $T_h = 473$ K and 673 K. It can be seen that the performance of 80-scale screen is the best, while the 180-scale screen is the worst among the three. The magnitudes of all quantities resulted from the 80-scale screen are about 20–30% larger than those by the 180-scale screen. The differences between the 80-scale screen and 120-scale screen are much smaller, but the former clearly performs better than the latter. The results seem to suggest that in this regenerator, the importance of smaller pressure drop outweighs that of a large heat transfer area. The reason could be due to the gap between the displacer and the displacer cylinder. This gap allows part of the working gas to flow forwards and backwards between the expansion and compression chambers without entering the regenerator that is housed inside the displacer. Large pressure drop inside the regenerator forces more working gas to flow through the gap and less to flow into the regenerator. As a result, less working gas exchanges heat with the regenerator material and the effectiveness of the regenerator is hampered. The width of the gap in the current engine is 2 mm. By reducing the gap width should improve the effectiveness of the regenerator and the performance of the engine because some working gas that just passing by through the gap before would be forced to flow into the regenerator. To prove this argument, case 9 is performed where all parameters are the same as those in case 4 except the displacer radius has been increased by 1 mm, that is from $R_d = 0.078$ m to $R_d = 0.079$ m. This reduces gap width from 2 mm to 1 mm. The results of case 9 are also plotted in Fig. 9. Compared with the results in case 4, it is clear that reducing the width of the gap has yielded marked improvement on engine performance. Here, engine's shaft torque, power, and efficiency have been increased by 10–20%. This has proven the argument on the effect of narrowing the gap between the displacer and displacer cylinder and clearly demonstrates that the physical mechanism for better performance in this group of cases is allowing more gas to flow into the regenerator rather than just passing by the regenerator through the gap between displacer and displacer cylinder. This can be achieved by decreasing the pressure drop inside the regenerator or narrowing the gap. However, further attempt to reduce this gap has resulted in physical contact between the displacer and the inner wall of displacer cylinder during the rapid reciprocation motion of the displacer; hence, the results will not be accurate due to additional friction introduced by the physical contact.

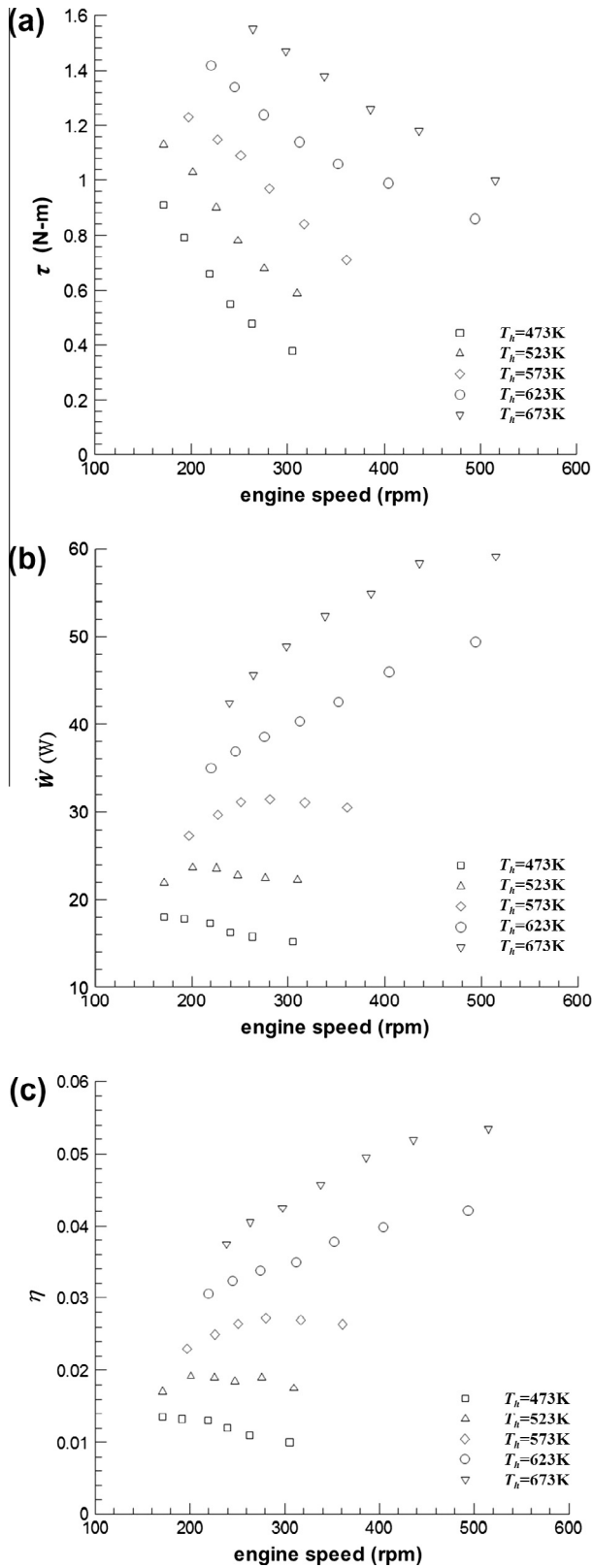


Fig. 6. Variations of engine shaft torque, power, and efficiency versus engine speed at 5 different temperatures in case 1; (a) engine shaft torque, (b) engine power, and (c) engine efficiency.

4.4. Effects of regenerator fill factor

Cases 7, 4, and 8 are designed to investigate the effects of fill factor. In these cases, 80-scale matrix screens are used, the masses

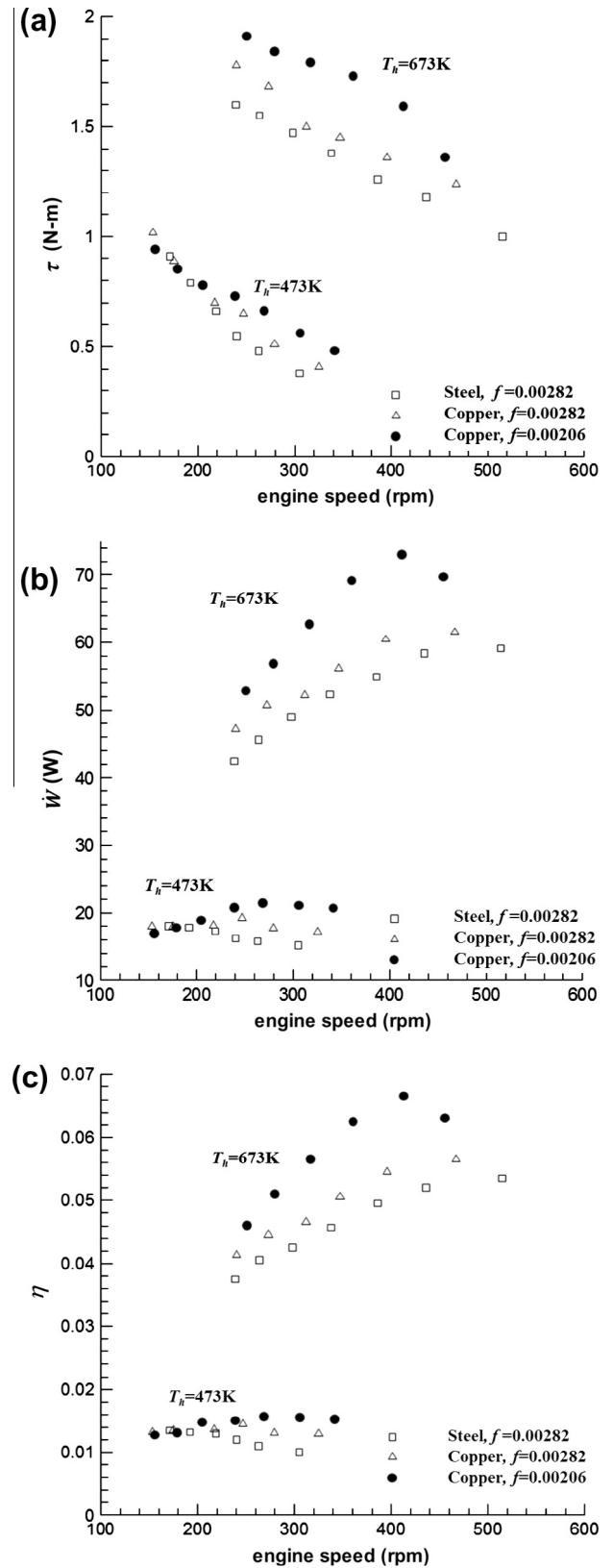


Fig. 7. Variations of engine shaft torque, power, and efficiency versus engine speed at 2 different temperatures in cases 1–3; (a) engine shaft torque, (b) engine power, and (c) engine efficiency.

of matrix material of them are 0.072, 0.1, and 0.121 kg, respectively, and the fill factors are 0.00183, 0.00282, and

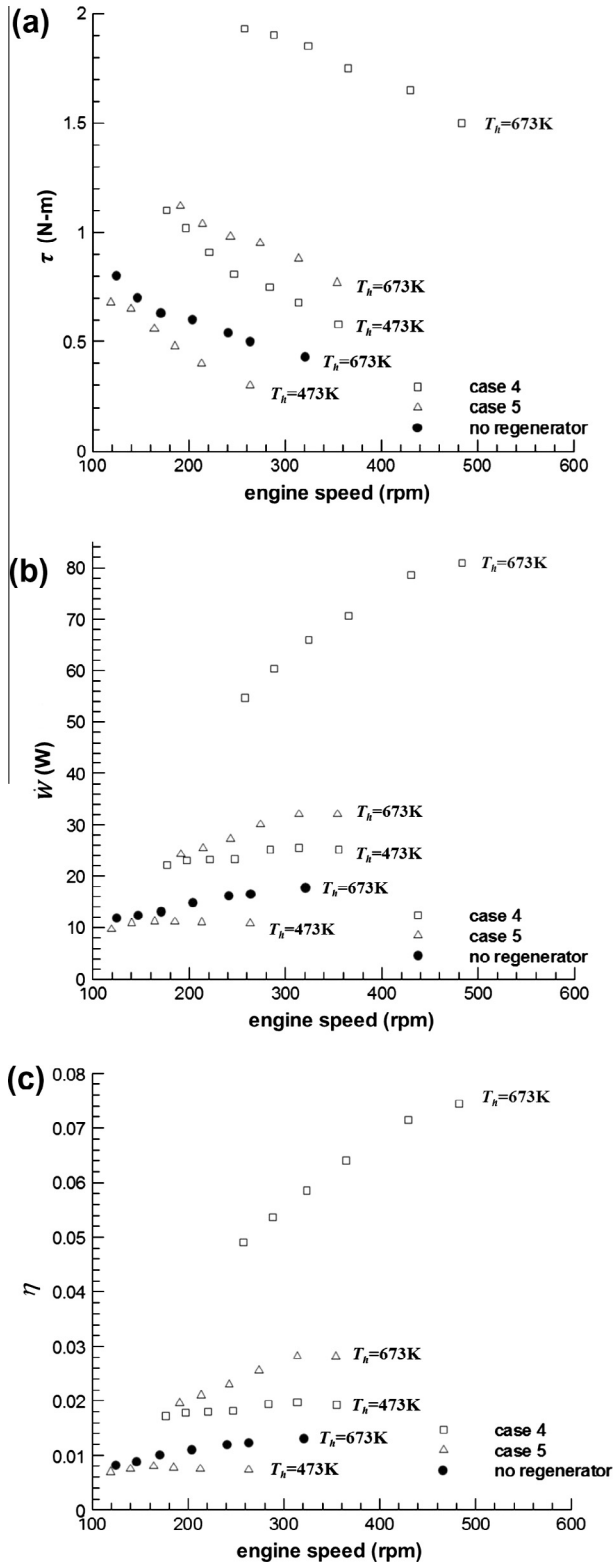


Fig. 8. Variations of engine shaft torque, power, and efficiency versus engine speed at 2 different temperatures in cases 4 and 5; (a) engine shaft torque, (b) engine power, and (c) engine efficiency.

0.00356, respectively. A larger fill factor introduces advantages of larger regenerator heat capacity and total heat transfer area but disadvantage of larger regenerator pressure drop, which limits working gas to flow into the regenerator to exchange heat with matrix material. The situation could lead to the existence of an

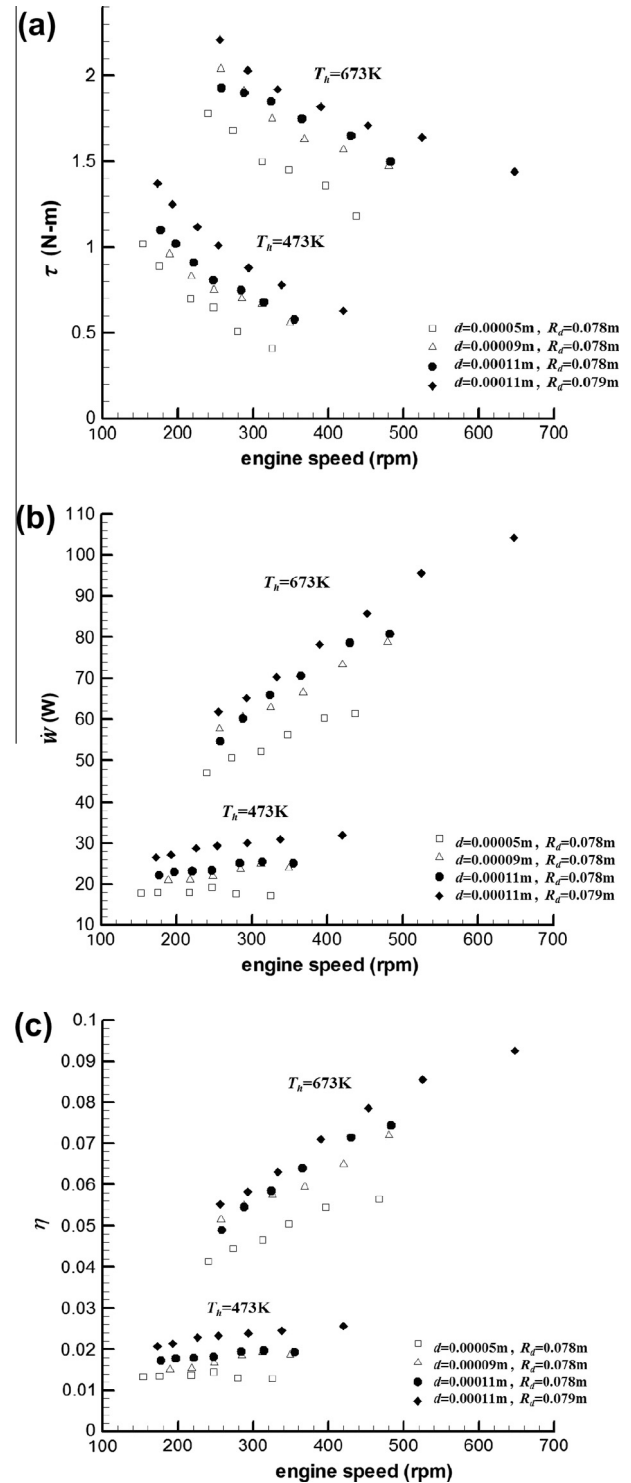


Fig. 9. Variations of engine shaft torque, power, and efficiency versus engine speed at 2 different temperatures in cases 2, 4, and 9; (a) engine shaft torque, (b) engine power, and (c) engine efficiency.

optimal fill factor which can achieve the best balance between the advantages and the disadvantages introduced by increasing fill factor. The variations of shaft torque, engine power, and efficiency versus engine speed of the three cases at two different hot-end temperatures, $T_h = 473\text{K}$ and 673K , are given in Fig. 10. The results indicate that case 4 ($f = 0.00282$) clearly performs the best among the three, while cases 7 and 8 have mixed performance, depending

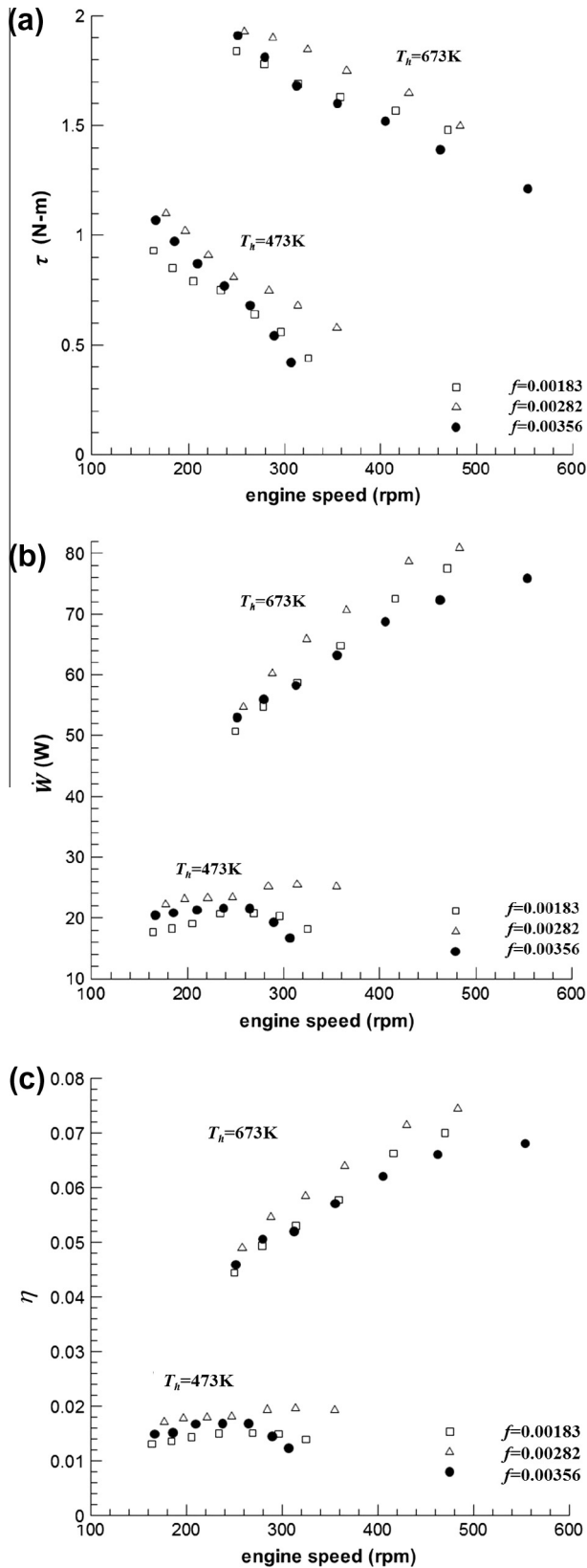


Fig. 10. Variations of engine shaft torque, power, and efficiency versus engine speed at 2 different temperatures in cases 7, 4, and 8; (a) engine shaft torque, (b) engine power, and (c) engine efficiency.

on the loading (speed) of the engine. At larger loading (lower engine speed) condition, case 8 performs better than case 7, producing larger shaft torque and engine power, whereas, case 7

performs better than case 8 at smaller loading (higher engine speed) condition. Such results suggest that an optimal fill factor does exist. The performance of cases 7 and 8 further indicates that a regenerator with larger fill factor only performs better at low engine speed. This suggests that large heat capacity and heat transfer area are more important at low engine speed but low regenerator pressure drop is more important at high engine speed. The physical mechanism contributing to better performance in this group of cases is the optimal combination of reducing regenerator pressure drop (by decreasing fill factor) and promoting heat capacity and heat transfer area (by increasing fill factor). Since an increase in fill factor can introduce both positive and negative effects on engine performance, it is very important to find the balance between the advantages and disadvantages introduced by increasing the fill factor according to the operational conditions of the engine.

5. Conclusions

A helium-charge γ -type twin-power piston Stirling engine has been built, and a test facility has been constructed to measure the engine's performance. The engine incorporates a moving regenerator which is housed inside its displacer. Stacked-woven metal screens have been used as the regenerator matrix material. The main contributions of this study are: first, it investigates the characteristics of a moving regenerator and the impacts of several regenerator parameters on engine performance, which have not yet been reported in such detail in open literature. Second, an innovative method of bracing regenerator matrices inside the moving regenerator to withstand rapid reciprocation motion has been proposed. Third, it identifies the importance of the arrangement of regenerator matrices in relation to the flow direction, a factor that has not been well recognized or studied before. The regenerator parameters investigated here include screen material, screen arrangement, screen wire diameter, and fill factor. The results include engine shaft torque, engine power, and efficiency versus engine speed. Some conclusions drawn from this investigation are summarized as follows:

1. Between the two test metals, copper, due to its much higher thermal conductivity, has been shown to be a superior material than stainless steel under the test conditions in this study.
2. The arrangement of matrix screens in the regenerator has been proven to be a very important factor affecting the overall performance of the engine. Two different ways on installing the matrix screens have been investigated. One is with these screens installed in several layers in a manner that the flow direction of the working gas is perpendicular to the screen surfaces, and the other is that the flow direction is parallel to the screen surfaces. The former installation returns much better engine performance than the latter, yielding 2.3–2.5 folds increase on engine shaft torque and power.
3. Three copper screens with different wire diameters have been examined. The results indicate that the largest wire diameter gives rise to the best engine performance, whereas, the smallest wire diameter gives the worst. Smaller wires produce a larger total heat transfer area between the working gas and matrix material, but they also create larger pressure drop inside the regenerator due to their tighter weave pattern. The increase on regenerator pressure drop drives more working gas to flow through the gap between displacer and displacer cylinder, thus decreases the effectiveness of the regenerator.

4. The performance of the engine can be further improved by increasing the radius of displacer, resulting in reduction of the gap between displacer and inner wall of displacer cylinder. The narrower gap directs more working gas to flow into the regenerator to exchange heat with the matrix material thus improves the effectiveness of the regenerator.
5. Three different fill factors have been examined. It is found the middle fill factor results in the best engine performance, indicating the existence of an optimal fill factor. The engine with the largest fill factor only performs better than the smallest fill factor at low engine speed. At higher engine speed, the latter performs better than the former. The mixed performance of large fill factor further proves the importance of finding the balance between the advantages and disadvantages introduced by increasing the fill factor.

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